TWELVE YEARS To Discover the Obvious!

Steps in the evolution of a variable speed transmission

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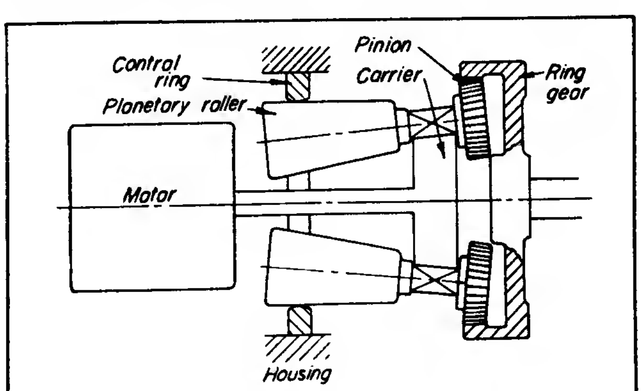
THIS article takes the reader behind the scenes and tells of the successive steps in reaching the solution to what would appear to be a relatively simple problem in machine design. As so often happens the answer, when finally arrived at after a series of five detours covering a period of twelve years, proved to be the simplest, most direct, most economical and—it must be admitted—the most obvious.

The problem concerned a single element in the design of a variable speed transmission using metallic rolling traction, *Fig.* 1. It was to find the best way to mount the tapered planetary rollers in their carrier so that they would contact the encircling ring and build up the tractional force required. To understand what was wanted, the operation of this transmission, as more fully described in the author's article, "Planetary Transmissions", MACHINE DESIGN, Nov., 1946, will be reviewed briefly.

This drive is the counterpart of the familiar compound differential planetary geared transmission, known to millions of early motorists in the Model T Ford, except that tapered rollers on inclined axes contacting an encircling traction ring replace the three sets of planetary pinions and contacting gears used in the Ford. It will be remembered that by selectively engaging the clutch or wrapping a brake band around drums fastened to two of the contacting gear sets, the Ford was put into high, low or reverse. In the Graham transmission, instead of three speeds an infinite number is had by moving the stationary contact ring axially along the outside of the planetary rollers (whose outer edge is parallel to the central axis, since the taper of the rollers equals their inclination) to engage an infinite number of different diameters of the rollers and so give all speeds from top to zero and reverse.

Calculation of the output speed in the Graham is similar to that in the Ford. At the small end of each of the planet rollers there is fastened a planet pinion which meshes with a mating ring gear keyed to the output shaft. The linear speed of this ring gear is the difference between the constant linear speed of the motor at the pitch line of the gearing minus the linear speed about its own axis of the planet pinion, which is made to rotate in the opposite direction from the motor by the friction or reaction of the tapered roller against the ring. As the con-

Fig. 1—Diagram showing the essential elements of variable speed transmission using metallic rolling traction



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trol ring is moved to the small end of the roller, the roller with its pinion turns faster in the ratio of the relative diameters of roller and ring bore until the speed of the pinion about its own axis equals its speed about the central axis, at which point the output speed becomes zero. Beyond this point, the output shaft reverses.

The problem is to so mount the rollers in their carrier or spider, *Fig.* 1, as to secure the traction needed to give the roller its planetary rotation. Incidentally, since this rotation is produced against the resistance at the teeth of the ring gear, arising from the external applied load, the numerical value of tangential tractional force required at each roller contact must equal the gear tooth pressure times the pitch radius of the planet pinion, divided by the radius of the roller at its contact point with the control ring. This reaction force, which again may be likened to the reaction between rail and driving wheel necessary to push a locomotive ahead against the drawbar pull and other resistances, is derived from the radial force between roller and ring, times the available coefficient of friction or traction. Since the transmission runs in oil, this coefficient is very much less than in a locomotive or automobile that uses dry traction, so that the radial force between roller and ring must be correspondingly greater. This radial force is had by pushing the roller against the ring by the centrifugal force due to its rotation, or by a spring, or by a torque-actuated cam or other device, or by a combination of these loading means. The roller must be so mounted in the carrier as to permit the development of such force, and herein—as already stated—lies the subject of this article.

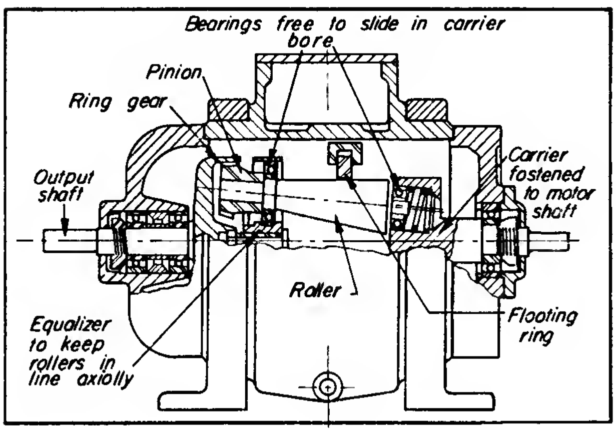


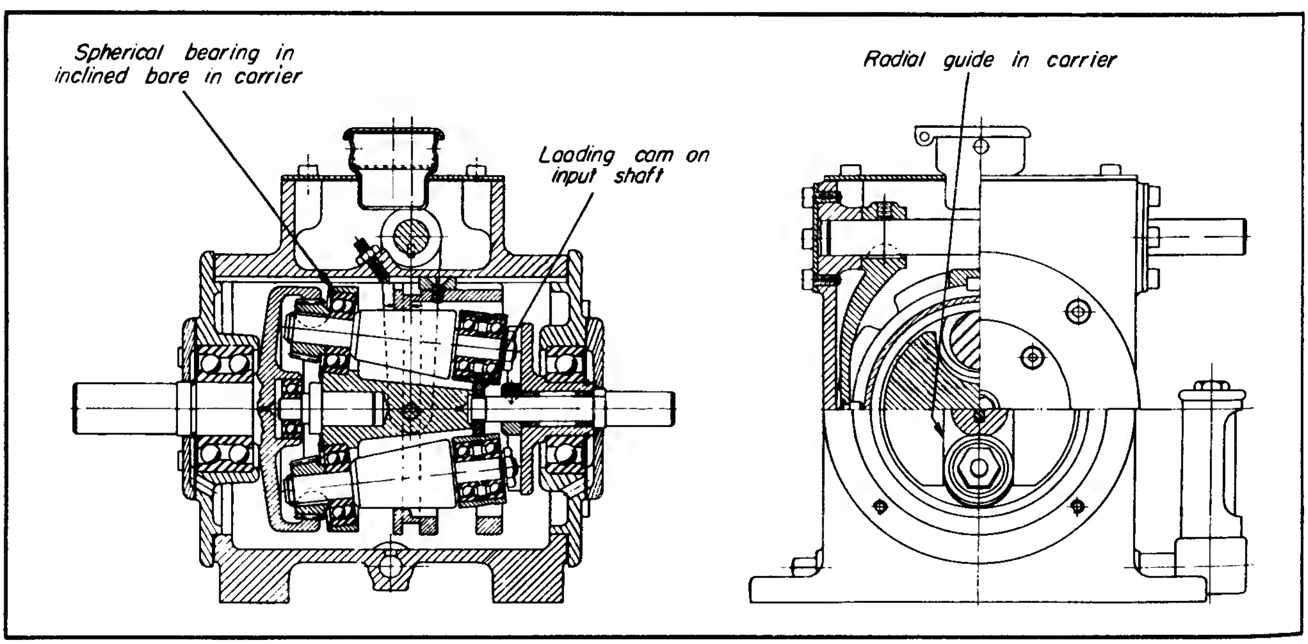
Fig. 2—Above—First design had bearings which were free to move outward in inclined bores in the carrier under the influence of centrifugal force and a spring

Fig. 3—Below—Second design employed torque-responsive loading applied through a cam on the input shaft

FIRST DESIGN: *Fig.* 2 shows the original mounting employed for this purpose (the drawing is reproduced from *Kent's Handbook,* 11th edition, 1936). Each of the three planetary rollers has a bearing at each end, loosely fitted in inclined bores in the carrier. When the spider rotates at motor speed, the centrifugal force acting on each roller urges the roller outward along its inclined axis.

Since the bearings of the roller are free to move up in the carrier bores, they will do so until the rollers contact the floating ring. Assuming that the centrifugal force is then completely absorbed by an equal and opposite force of contact at the ring, these two forces comprise a couple which may obviously be balanced in theory by a corresponding couple developed by the two bearing reactions, each of which then equals numerically the amount of the couple divided by the bearing span.

Unfortunately, the pitfall in this assumption is that conceivably one bearing could be omitted, in which case the centrifugal force (plus spring pressure back of the roller where used) could be fully sustained by the other bearing and the ring, in which event the bearing reaction, when the ring was adja-



cent to it, might be as much as eight times as great as if both bearings were operative. In other words, the bearing reactions were inherently indeterminate in this design and in practice the bearings proved inadequate since each should have been large enough (if space permitted) to carry the full load, in which case, of course, one of them might as well have been left out.

SECOND DESIGN: This was, in effect, done in the second design, *Fig.* 3, as far as supporting the centrifugal force was concerned. The bearing at the larger end of the roller was now carried in a slide in the carrier which evidently provided no outward reaction, the slide merely serving to locate the roller tangentially. This tangential location was necessary since a single fixed spherical bearing was substituted in this design for the two outwardly sliding bearings, the roller now being free to swing radially, on the bearing center as a pivot, into contact with the ring.

An incidental advantage of this mounting was that each roller moved independently of the others and the ring, instead of floating, could be given a minimum sliding clearance in the housing; whereas in the previous design, in addition to the floating ring, an equalizing device was necessary to keep the rollers in step axially as they moved up into the ring bore.

This second design also introduced a new method of loading the rollers to augment and largely replace the centrifugal force. It was considered an advantage to have the loading force increase with increase in load so as (1) to relieve the contact pressure when the external load was light, and (2) to prevent slippage when the load became heavy.

Although this idea of "torque-responsive" loading is attractive in theory there turned out to be two good reasons against its use in the Graham. First, in the Graham a complete slip at the roller contact is impossible even in the case of a complete stall of the driven shaft because the roller will continue to roll within the ring, even though the output shaft is held against rotation. Thus, torque responsive loading is not necessary in the Graham to prevent destructive slippage as with other devices, and actually the Graham serves in many applications as a form of load limiting clutch to safeguard the driven machine and transmission against dangerous momentary overloads which are inevitable in certain types of equipment such as conveyors. But the second and more important reason for the abandonment of torque loading is that in case of a jam at low speeds, where the motor protective devices do not function, the forces in the transmission itself may then reach destructive values and thus do as much harm as the very slippage which this type of loading aims to avoid.

THIRD DESIGN: Accordingly, in the third design,

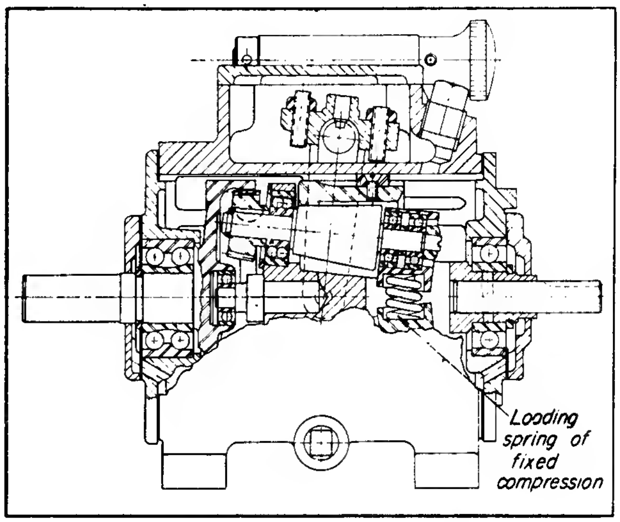
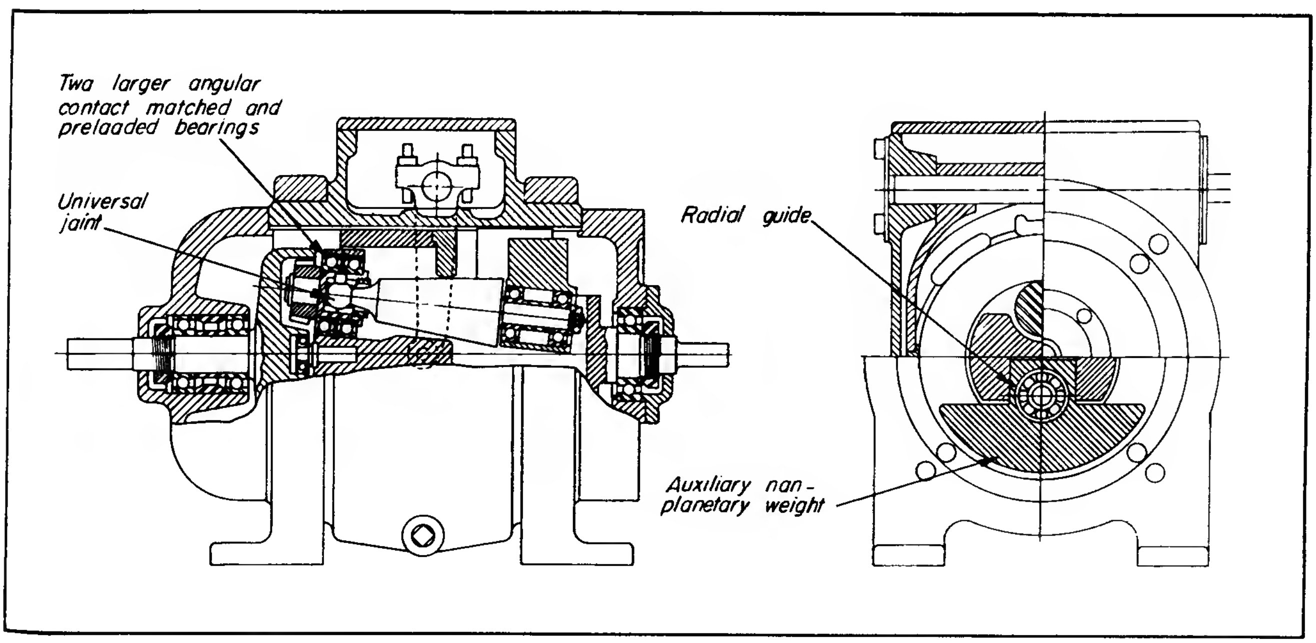


Fig. 4—Above—Third design had fixed instead of torque responsive loading, by spring plus centrifugal force

Fig. 5—Below—Fourth design had special universal joint at small end of roller instead of spherical bearing. Auxiliary weight provided additional centrifugal force



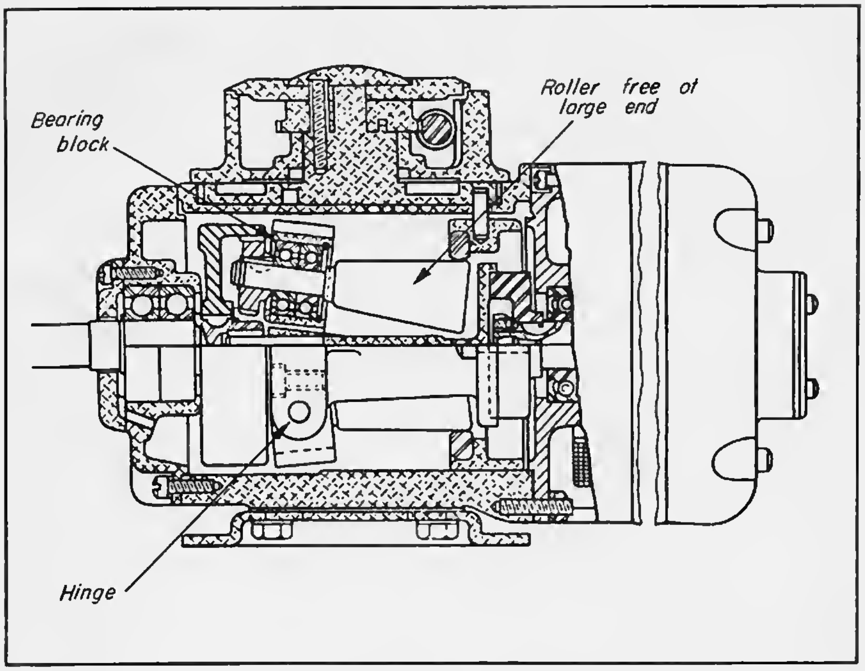


Fig. 6—Left—In fifth design rollers were free at their large end and carried in hinge-mounted double ball bearings at small end

Fig. 7—Below—Roller and carrier assembly for fifth design, showing hinge mounting to permit rollers to move out into contact with the ring

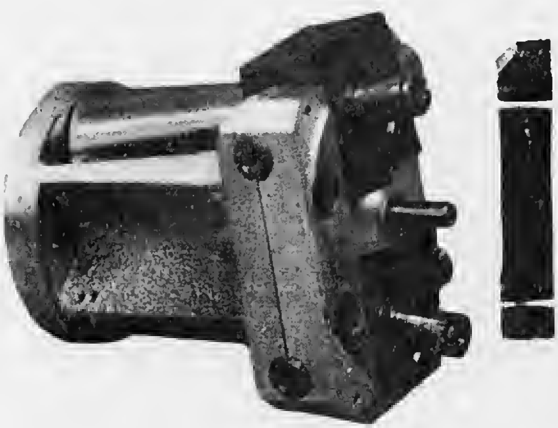


Fig. 4, the cam on the input shaft which applied a contact pressure between roller and ring proportional to the input torque, was replaced by a simple pillow block to give the desired loading through a *fixed* amount of centrifugal force plus, in some cases, an auxiliary spring. This had a further advantage over the previous torque-responsive loading in that it nearly doubled the torque capacity at low speeds (near zero) as compared to high, which is desirable in nearly all applications of variable speed drives whereas with the input cam loading the torque capacity decreased at low speeds, for reasons which need not be here explained, but are apparent by analysis of the mechanics involved.

This third design, with the roller carried by a fixed spherical bearing at its small end and a pillow block sliding in a guide in the carrier at its large end, appeared in units that were widely used during the war. Thousands were employed for the tracker drive on gun directors, airplane cameras, reproduction equipment, machine tool feeds, etc.

FOURTH DESIGN: But even this design, contrary to the popular delusion that inventions may spring Minerva-like in final form from their creator's cranium, was found to have one serious limitation or Achilles' heel (to continue the mythological allusion) that greatly curtailed its application. For a given dimension of housing, or ring bore, which establishes the size of the transmission, the rating was determined and restricted by the capacity of the spherical bearing. This is because the strength of the rollers and ring was inherently in excess of that of the spherical bearing, whereas an economical design, as every engineer knows, must approach the ideal of the "one hoss shay." Obviously, two bearings instead of the one could not be used here without abandoning the pivot. Therefore the next step, which required three years to perfect, was to work out a special and unique form of universal joint, *Fig.* 5, to replace the spherical bearing and permit the use of two larger, angular contact preloaded bearings whose capacity—particularly in thrust—was several times that of the single spherical bearing. In addition, the sliding pillow block was weighted at its outer surface to add to the centrifugal force.

These seemingly simple changes greatly extended the field of application of the drive by increasing the capacity for a given size. Conceivably the problem could then have been considered solved, except for the fact that the universal joint was an expensive part to make and in connection with the weighted pillow block at the large end of the roller still kept the cost per delivered horsepower at too high a figure.

FIFTH DESIGN: The next step was the discovery of a way to eliminate both the joint and the pillow block, decrease the number of parts, particularly the number of ball bearings, and add one-third to the capacity for a given size. All this cut the cost of the transmission by nearly one-half and was accomplished

(see *Fig.* 6) by replacing the universal joint with a "hinge" construction and eliminating the guide at the large end of the roller entirely. The omission of this guide was possible (this had not previously been realized) because the available traction coefficient in a lubricated drive is so low as to permit the tangential reaction at the ring contact to be fully supported, cantilever-fashion, by the same two bearings at the pivot, which share the radial load with the ring.

Instead of the universal joint, then, a bearing block carrying the two angular-contact, matched and preloaded bearings was hinged in the carrier, and since there was now no block at the large end of the roller, this not only eliminated two bearings for each roller at that point but added one-third to the effective length of the roller in a given housing, which in turn added one-third to the capacity. All this meant more power with fewer parts and less cost—a real advance.

Final Step Brought Biggest Gains

Now at last the designers might have been content to rest on their laurels. But the real denouement was yet to come, which was to add nearly 50 per cent more (on top of the previous gains) to the capacity of a given size transmission, replace a seven-piece roller carrier assembly by a single-piece carrier, give greater stability, quieter operation and still greater accuracy of speed holding and speed setting.

Final Design: Up to this point all devices of this general type (and the patent files show a large list) had based the design of the roller mounting on the assumption, apparently a quite natural one, that in order for the roller to make pressure contact with the ring, it would have to be "flexibly" mounted in the carrier—free to "move up", so to speak. Hence, the use of axially movable bearings in the Graham first design, the spherical bearing in the second and third, the universal joint in the fourth and the hinge in the fifth. This flexibility in each case meant added structure, added parts, added cost—*and actually wasn't needed at all.*

It has already been explained that the radial centrifugal force of the roller in any event divides between the bearings at the small end of the roller and the ring (in proportion to the distance of the center of gravity of the roller assembly from the bearings and ring respectively). The ring, therefore, is really the counterpart of a bearing at the end of a shaft—but full use had not previously been made of this important fact.

It was finally realized that if the rollers could in the initial assembly be so positioned that each was in contact with the ring bore (no matter what the axial position of the ring), then the bearing at the small end of the roller could be locked in position in the carrier. Then, when rotation started, the centrifugal force would divide between bearings and ring as above explained, just as in the other designs. The initial positioning required merely a few shims of required thickness (see *Fig.* 8) inserted between the bearings and a shoulder in the inclined bore of the carrier. Yes, a few shims costing less than a cent took the place of joints, hinges, blocks, etc., and at the same time permitted a simple one-piece carrier to do the job of mounting. Too, this elimination of parts enabled a third roller to be introduced into the same size casting that previously could house only two, adding 50 per cent to the capacity of a given size transmission. Incidentally, the third roller gave other important advantages of improved quiet, still closer speed holding and speed setting, etc. But the big achievement was simplicity, low cost, and of course the added reliability that goes with fewer parts.

Here then was the obvious way to mount planetary rollers in their carrier in a variable speed transmission of the tractional type. But why the five detours?

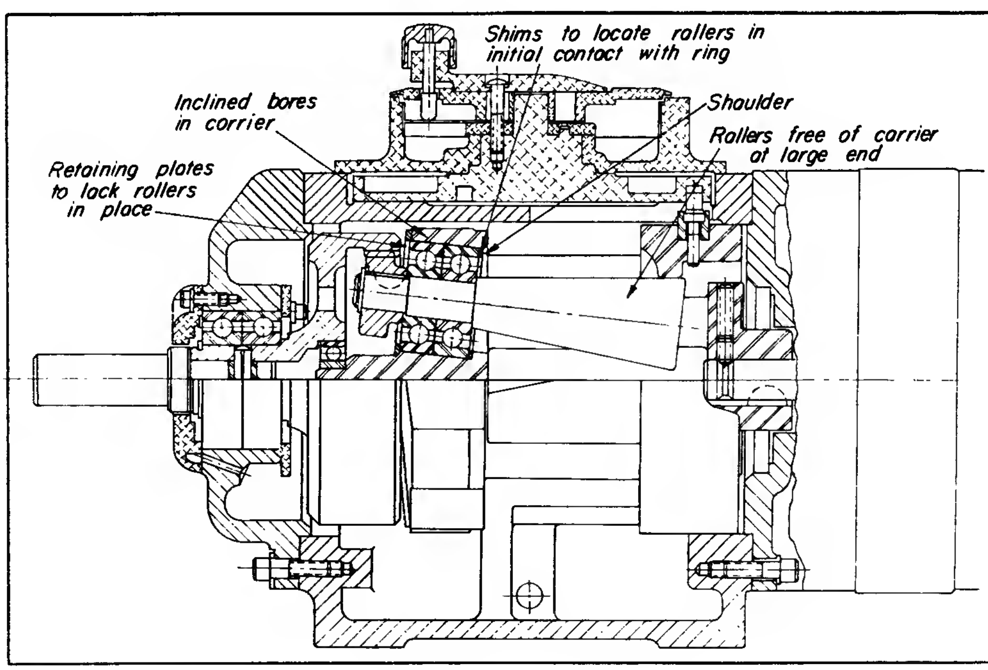


Fig. 8—In final design rollers are free at their large end. Bearings at small end are locked in position at assembly, after rollers have been brought into initial contact with ring by shims inserted in carrier